HYDRAULIC DRIVE SYSTEM

FIELD OF THE INVENTION:

The present invention relates to a hydraulic propulsion system for a human powered vehicle of the type in which the vehicle operator provides the power for propulsion by charging an accumulator and controlling flow of hydraulic fluid to one or more motors.

BACKGROUND TO THE INVENTION:

There is a need for a human powered urban vehicle as an alternative vehicle that fills the gap between the bicycle and the automobile. Such a vehicle is of great utility in urban and third world environments where, due to population and space reasons, automobiles are not universally utilized. In such environments there is also a consumer demand for an alternative that provides a good ride, weather protection, and ease of use in crowded cities.

In the past, many attempts to replace the bicycle have resulted in vehicles which suffer from numerous disadvantages. So called recumbent bicycles wherein the rider is seated in a position part way between sitting upright and reclining have disadvantages in that the very low seated position of such vehicles reduces visibility and increases the likelihood of accident or injury. Such vehicles have not gained acceptance among conventional bicycle riders and therefore have a very limited market.

A preferred type of human powered vehicle features a chassis on which are mounted three independently sprung wheels, two in the rear and one steerable front wheel. The suspension allows the vehicle to lean like a bicycle around corners, yet remains upright when stopped or moving slowly. The vehicle avoids a low seating position. Careful design has also facilitated a number of features found to be desirable in such vehicles through a consumer preference survey. Such features

include a comfortable seat. The rider's legs project somewhat forward, allowing a bucket-type seat to be used, which allows the rider to push on pedals as hard as desired against the seat back rather than being limited by his body weight. Hill climbing is accordingly easier. A step through frame is also provided so the rider can easily mount and dismount. Weather protection is also provided by an optional weather protection shell which completely protects the rider from rain and cold when needed. Equally a rear carrier storage area is also protected to keep the cargo dry. An enclosed carrier may be located behind the seat and between the rear wheels easily to carry packages. Carrying loads at a lower level improves stability of the vehicle when carrying substantial loads.

SUMMARY OF THE INVENTION

The present invention provides a hydraulically operated propulsion system for such a vehicle, replacing conventional chain and sprocket drives currently used in bicycles, which system has numerous advantages over conventional bicycle operating mechanisms. The drive system is inherently low maintenance, since all drive components are sealed in oil. Further, a hydraulic drive system includes an accumulator in which pressurized oil may be stored which allows car-like acceleration when required. During braking, the hydraulic drive system has the ability to capture otherwise wasted brake energy by pumping up the pressure in the accumulator, thus allowing the vehicle start up again from stop without pedaling.

In accordance with an embodiment of the invention, a hydraulic propulsion system for a human powered vehicle comprises: a reservoir, a treadle pump having dual treadles, an accumulator, a hydraulic variable ratio pressure transducer, and at least one hydraulic motor. The reservoir is connected to the pump, the pump is connected to the accumulator, or motor and valve means connects the accumulator to the motor.

The hydraulic variable ratio pressure transducer is in circuit between the accumulator and the motor. The motor is connected to said reservoir, also pressurized hydraulic fluid is supplied to the accumulator from the pump, and pressurized hydraulic fluid is provided from the accumulator to the motor, and hydraulic fluid is returned to the reservoir from the motor or to the variable pressure ratio transducer.

In accordance with another embodiment, the variable pressure ratio hydraulic transducer comprises a vane type armature which freely rotates within a cam ring. The armature includes a radially extending passage intermediate each pair of vanes, the armature rotates about a porting ring having ports for communication with the armature passages, and including a centrally mounted control vane having passage means therein, port means for connects the transducer to a source of pressure and to an outlet in pressure ratio relationship to the source of pressure. Means are provided for manually controlling the position of the control vane for varying the pressure ratio of the transducer.

BRIEF DESCRIPTION OF THE DRAWINGS

A better understanding of the invention will be obtained by a consideration of the detailed description below of a preferred embodiment, with reference to the following drawings, in which:

Figure 1 is a simplified view of a hydraulically operated human powered vehicle of the invention;

Figure 2 is a schematic diagram of the hydraulic circuit of the vehicle;

Figure 3 is a physical layout illustrating the controls and principle components of the hydraulic propulsion system;

Figure 4 is a schematic view of one side of the treadle operated hydraulic pump of the hydraulic system;

Figure 5 is a top view of the pump with the treadles removed;

Figure 6 is a side view of the linkage for controlling the variable stroke connection between the treadle and connecting rod of the piston;

Figure 7 is a section through a wheel motor in accordance with the present invention;

Figure 8 is a mechanical schematic of the bidirectional clutch, shown linearly for simplicity;

Figure 9 is a detail of Figure 8;

Figure 10 is a view on the line 8-8 of Figure 7;

Figure 11 is a cross-section through the control valve of Figure 2;

Figure 12 is a cross-section through the 1, 2, 3 valve of Figure 2;

Figure 13 is a schematic diagram of the valve 24 of Figure 2;

Figure 14 is a section through the hydraulic variable transducer of Figure 2;

Figure 15 is a plan view of the porting ring of Figure 14;

Figure 16 is an elevational view of the porting ring of Figure 14;

Figure 17 is a plan view of the control vane of Figure 14;

Figure 18 is an elevation view of the control vane of Figure 14;

Figure 19 is an axial view of the armature of the pressure transducer;

Figure 20 is an end view of the armature of Figure 19;

Figure 21 is a plan view of an end plate of the transducer of Figure 14;

Figure 22 is a schematic view of a four chamber transducer, and

Figure 23 is a view of the central control shaft and porting of Figure 22.

DETAILED DESCRIPTION OF THE DRAWINGS:

Figure 1 illustrates a human power driven vehicle incorporating continuous power input and energy storage and recapture via a self-contained hydraulic system having an accumulator, a constant displacement vane motor, a variable stroke piston pump, and other components. The vehicle 10 consists of a chassis 11, a steerable front wheel 12 and a pair of independently suspended rear wheels 13. A seat 14 is provided for the rider, and a steering mechanism 15 operated by the rider controls the position of the independently sprung front wheel.

The rider is positioned in a semi-upright position with hands grasping steering handlebars 15 and feet positioned on treadle pedals 16. Treadle pedals 16 operate a pair of variable stroke, variable displacement piston pumps 20 as a component of the hydraulic propulsion system.

Referring to Figure 2, the vehicle power input consists of suitable exertion by the operator against the treadles which operate the variable stroke piston pumps 20. Dynamically, variable stroke length is accomplished by a manually controlled linkage described hereinafter connected to conventional piston pumps. Check valves (not shown) are included in the pumps in the normal fashion for pumping hydraulic fluid from a low pressure reservoir 21 to the high pressure side 22 of the hydraulic circuit. In a preferred embodiment, two hydraulic motors 30 are provided at the rear of the vehicle, each driving one rear wheel. These motors are constant displacement vane type motors which operate as a motor upon start up and during cruise operation and can be operated as pumps to provide regenerative braking during deceleration. A control system is provided as discussed hereinafter which controls operation of the vehicle during all phases from stationary to accelerating, constant speed, decelerating, and braking. The vehicle is also provided with secondary brakes which are preferably internal

expanding drum brakes to supplement the regenerative braking when required.

In accordance with the invention a hydraulic variable pressure ratio transducer 25 is included in the hydraulic circuit which greatly improves the efficiency of vehicle operation.

Referring to Figure 2, there is disclosed schematically a hydraulic operating system of the vehicle of Figure 1. A double piston treadle pump 20 draws fluid from a reservoir 21 and delivers this fluid to the high pressure line 22. There the fluid is subsequently routed to valve 24 or valve 23. Valve 24 feeds the hydraulic fluid to a hydraulic variable pressure ratio transducer 25 and through a control valve 26 to an accumulator 27. Consequently, the wheel motors 30 can receive fluid under pressure from the transducer 25 via the valve 23,24, and 26 directly from the treadle pump 20 via valve 23, or any desirable combination thereof. Under the regenerative braking, all of the flow from the wheel motors 30 is combined with any oil flow in pipe 22 and passes via valve 24 to the transducer 25. Relief valves 28 and 29 return fluid from the high pressure side of the hydraulic circuit to the reservoir 21 in the event there is an over-pressure in the high side of the hydraulic circuit. Consequently, wheel motors 30 under the control of valve 23 provide propulsion to the vehicle and during regenerative braking charge the accumulator 27.

Reservoir 21 may preferably be an air-over-oil reservoir provided with a diaphragm to retain cleanliness in the hydraulic circuit and accumulator 27 may also be an air-over-oil accumulator with a diaphragm.

As mentioned above, the two wheel motors 30 are provided which are constant displacement vane motors that can also be reversed to pump fluid to the accumulator 27 for regenerative braking. The motors 30 are mounted coaxially with the wheels and the motor bearings also serve as wheel bearings. All

energy to and from the wheels passes through the wheel motors and there are no chains, gears, belts, sprockets or other mechanical drive components. Details of the individual wheel motors will be discussed below in relation to Figures 7 and 10. Each of these motors 30 is also provided with a bi-directional clutch which is integrated with the wheel motor that serves to disconnect the motor armature from the wheel shaft and removes drag when coasting and allows the vehicle to achieve a coasting performance comparable with a bicycle. The clutch re-engages automatically when the motor 30 is driving a wheel, and is manually engaged for regenerative braking.

The accumulator 27 is a pressure storage system of about 4 litres in volume which uses the spring force of compressed air behind a diaphragm to store energy in the form of pressurized oil. Either by itself or in combination with the treadle pump, it allows the delivery of energy at much higher rates than the rider can deliver, facilitating rapid acceleration. For deceleration, it can accumulate braking energy regeneratively. Control valve 26 controls oil flow to the accumulator and to the transducer 25. The control valve is intended to prevent or extend the leak down of the accumulator and is shown in Figure 11. The control valve 26 can also serve as a flow limiter or governor for wheel speed.

Figure 3 which is a physical layout illustrating the controls and principles of the hydraulic propulsion system shows the piston pump 20, drawing oil from the reservoir 21 and delivering pressurized oil to the high pressure side of the hydraulic system 22. Pressurized oil is then directed by the energy control module 34, either to the accumulator 27 or the wheel motors 30. The energy control module 34 includes valve mechanisms 23, 24 and 26 (Fig.2) together with a hydraulic variable pressure ratio transducer 25. Right and left hand grips 31 and 32 which are of a conventional rotary type well known in the field of motorcycle controls and brake levers 35

and 33 are provided on the handlebars (not shown). The right-hand grip controls forward acceleration (f.a.), detent center (d.c.) and regenerative braking (r.g.). The left-hand grip 32 controls pedal displacement in the treadle pump 20 as disclosed hereinafter. The energy control module 34 also receives motor disconnect control m.d. from the left-hand grip 32 mounted lever and forward acceleration, detent center, and regenerative braking control from the right-hand grip 31. The energy control module 34 in turn controls regenerative braking clutches of the wheel motor 30, as will be detailed hereinafter.

When the right-hand grip 31 is in the detent center (d.c.) position operation of the pedal pump 20 can be used to pressurize wheel motors and propel the vehicle. Turning the right-hand grip 31 in the forward acceleration (f.a.) position will connect the accumulator 27 to the wheel motors 30 providing acceleration to the vehicle and if wanted in excess of the acceleration capabilities of a convention bicycle, and more nearly in line with the capabilities of an internal combustion engine automobile. Sustained pumping of the pump 20 will continue to provide hydraulic fluid to the accumulator and to the wheel motors as necessary. Turning the right-hand control 31 to the regenerative braking (r.g.) position will actuate the regenerative braking clutches of the motors 30, and changes valving causing the motors to deliver oil to the accumulator 27. As shown in Figure 2, relief valves 28 and 29 prevent over pressure in the high side of the hydraulic system or in the accumulator 27.

The details of the pedal pump 20 are shown in Figures 4, 5, and 6 of the drawings. In order to provide the best pump for the hydraulically operated vehicle, several factors were considered: (1) cost effectiveness, (2) efficiency, (3) variable displacement, (4) treadle design for reducing area required by the feet and reduce height knees had to come,

ground clearance, and at the same time allowing for partial treadle strokes, and (5) low force needed to change displacement of pump.

In accordance with the invention, it is important to avoid a rotary pump with crank pedals, such as conventional gear, vane, or axial piston pumps would constitute, as this would provide even resistance to rotation throughout the entire rotation when the accumulator is engaged with the circuit, even when the pedals are in the top and bottom dead center positions. In a conventional bicycle, one can exert as much or as little force as desired during pedal crank rotation, since the pedal crank will rotate easily as long as the bicycle is in motion. This allows the pedals to pass bottom dead center and top dead center easily where it is difficult to apply much force. A constant resistance rotary pump would be difficult to get past these points if the accumulator was engaged in the circuit, particularly when pedaling at high force.

The near linear motion of a treadle pedal is well suited to avoiding this problem. A rotary pedal is not suitable unless the problems discussed above are avoided. The treadle pump shown in the drawings is, of course, the preferred pump for operating the hydraulic system.

Referring to Figure 4, each drive arc 40 is pivoted on pivot 41 and is provided with an arcuate surface 42 having serrations for connecting to the foot 46 on the connecting rod 43 of the piston pump 44. Foot pressure on the arc 40 causes the arc to move in the direction to drive the connecting rod 43 and cause the piston 45 to pressurize oil in the cylinder 44 which is subsequently pumped to the high side 22 of the hydraulic circuit. The foot 46 on the connecting rod 43 is connected to the arc 40 by the interlocking teeth illustrated in Figure 4. The curvature of the surface 42 and of the foot 46 have the same center of curvature which is the pin 47. Both

the foot 46 and the surface 42 have interlocking teeth capable of preventing slippage under load.

The arc 40 is shown at the end of travel, and at this point a small gap is created between the foot 46 and the surface 42 due to a piston travel stop, which may provided for example by a pin 48 reaching the end of the slot in arm 49 as shown in Figure 5. It will also be appreciated that a piston stop can also be built into the end of the cylinder bore. Foot 46 can pivot around pin 48 but a detent-to-center spring and ball tries to place it as shown to match its arc with surface 42. Slotted arm 49 holds con rod 43 in its position by wrist pin 47. Slotted arm 49 pivots around pin 50 which is shown concentric with wrist pin 47 when piston 45 is at the bottom of its stroke. This slotted arm also has a detent-to-center mechanism, not shown, which the operator can reposition. Thus if the pump 20 were operating in the configuration shown and the operator wished to increase the displacement, the slotted arm detent mechanism would be shifted. When treadle 40 reaches the bottom of its stroke, the slotted arm 49 will move the connecting rod 43 and foot 46 to its new position. The foot 46 detent-to-center will ensure that the interlocking teeth on the foot 46 and the surface 42 match in order to accomplish this. While a single pedal 40 has been shown, a two cylinder pump with two pairs of slotted arms 49 would be present, one for each of the two cylinders. In a two cylinder pump, each pedal is connected to drive one piston and an interconnecting mechanism, not shown, can be provided to ensure that pedal movements are equal and opposite.

The piston return spring 51 and both detent-to-center mechanisms resist pedal force during travel. These are steel springs and efficiently return this energy later in the stroke.

It should be noted that the operator must use full strokes in order to effect a displacement change. When partial stroking occurs, which a treadle design can allow, the foot 46

never leaves the surface 42 and the slotted arm 49 cannot effect movement.

Oil enters and leaves the cylinders 44 by check valves which are self-actuating which eliminates the need for valve control mechanisms. Such check valves are inexpensive and prevent any downstream feedback from the rest of the circuit affecting pump operation.

Figure 7 is an axial section of a wheel motor 30. The motor consists of a shaft 60 having a distal end 61 provided with a threaded hole for securing a wheel thereto. The shaft 60 is mounted on bearings 67 and 62, bearing 67 being a roller bearing and 62 a ball bearing. A clutch pack 63 is fixed to the end of shaft 61 and connects torque tube 64 to shaft 61. The wheel motor rotor 68 is fixed to torque tube 64 and rotates therewith. An oil seal 65 is provided at the wheel end of shaft 60 in accordance with normal hydraulic motor construction. Wheel motor rotor 68 revolves on torque tube 64 and interfaces with cam ring 66 as detailed in Figure 10.

For the sake of simplicity, hydraulic connections to the motor have not been illustrated, but these will be evident to persons skilled in the art from the description which follows.

The motor is completed by end plates 69 and 70, between which the steel cam ring 66 is fastened.

It will be noted that the shaft 60 would normally be formed from mild stainless steel, the end plates 69 and 70 would be formed from aluminum, and the cam ring 66 and rotor 68 would be formed from steel. A single bolt and bevel gear, spline, or other type of keying can be used for fastening the wheel, and a single bolt provides for relatively rapid wheel change.

It should be noted that rotor 68 does not ride on the wheel shaft 60 but rather rides on the torque tube 64, which is connected by the clutch 63 during operation of the vehicle as detailed below. This arrangement completely isolates the

rotor 68 from forces arising from shaft 60 moving during cornering loads and flexing due to cantilever loads. Ball bearing 62 is provided to counteract thrust on the wheel shaft 60.

Figure 10 is a cross-section on the line 8-8 on Figure 7. The rotor 68 is shown in the steel cam ring 66. The rotor 68 is connected to the torque tube 64, for example by a crown spline 30% of rotor width. As illustrated, the rotor 68 includes a plurality of slots 71 in which are positioned pairs of pressure balanced dual vanes, and the cam ring 66 is provided with constant radius sections 72 and 73. Accordingly vane slot losses due to friction are eliminated by the constant radius sections in areas where vanes have a differential pressure across them. By virtue of the lower pressure design, small clearances and wide area of rotor end eliminates the need for pressure balanced end plates. It should also be noted that this design can make use of a constant radius section due to its relatively low speed. Conventional pump and rotor designs usually make use of constant vane acceleration profiles in order to allow higher rpm. Typical characteristics of the wheel motor for a one person vehicle would be 34.41 ml/rev. and at a pressure of 68.05 atm, 237.27 Newton-metres rev. or 37.76 Newton-metres torque with a 7.62 cm rotor with 1.91 by 0.366 cm travel vane.

Operation of the bidirectional clutch 63 of Figure 7 is explained below in relation to Figures 8 and 9. In order to explain the operation of the bidirectional clutch, it is shown for simplicity sake as a linear arrangement rather than as a circular arrangement. However the principles of operation are defined. Referring to Figure 8 and Figure 9, a grooved roller 80 occupies most of the pockets 81 in race 82. Springs 83 in slot 84 push rollers 80 to center bottom of pockets 81. This action moves retainer 85 to the position shown in Figure 9. Ratchet pawl 86 is attached to retainer 85 by a slot 87, and

is pushed to the right as illustrated in Figure 8 in the slot by springs (not shown). In the "passive" direction, the pawl 86 is spring loaded to enter pockets 88 in race 89 as race 89 travels in the direction of arrow 89A. During regenerative braking, another pocket 81 contains another pawl (not shown) which points in the opposite direction. Means are also provided for raising the pawl manually. Thus regenerative braking can be disabled if not required. The regenerative braking pawl would not be raisable while under load, but a brief motor direction pulse will cause it to release. If the race 89 moves to the direction of arrow 89A, pawl 86 will engage pocket 88 and cause retainer 85 to also move left. Slot spring for pawl 86 is strong enough to override all the roller springs 83 at once. This action also moves rollers 80 to the left and out into pockets 88 and 81. Roller pin slots in retainer 85 permit this. Rollers engage pockets 88 and 81 simultaneously transmitting the load. Pawl slot 87 and springs (not shown) prevent any significant load from being transmitted by pawl 86. Similarly when the reverse pawl is lowered the clutch will engage in the opposite direction. This type of clutch could also be built with two active pawls or two passive ones. A clutch with two passive pawls however would serve no purpose since it would resemble a solid connection and function. The groove in rollers 80 could be eliminated by providing a second spring slot in race 89.

This clutch has several benefits. First of all, the rollers 80 have a large load carrying capacity combined with the light engagement pawl 86, means that the clutch can be small as well as having a small coasting friction. The simple parts are easy to construct or obtain. Rollers for instance can be modified rollers from roller bearings. In addition, since the rpm of the wheels of the vehicle is relatively low, the small size of the clutch permits engagement of the regenerative braking without damage to the clutch.

Figure 11 is a section through the control valve 26 of Figure 2. This control valve is a pilot operated valve that has three purposes, firstly to prevent accumulator leak down. Without valve 26 the accumulator could leak down in as little as 10 minutes, but with the valve in the circuit the charge in the accumulator could last up to a month. This valve also prevents unintended reverse operation of the vehicle and prevents accidental over speed of wheel motors when in the free wheel mode or where a governor is present prevents excess in wheel speed in drive.

The valve 26 as illustrated includes a port A connected to the accumulator, a port B connected to the motors and a port R connected to the reservoir. The special features of the valve are illustrated in words on the drawing. During operation the valve has four possible conditions as follows:

1. The vehicle is stopped, the accumulator is charged and the brake signal is effected.

There is no pilot operation and the valve remains closed.

2. The vehicle is stopped or moving, the accumulator is charged.

A set of pilot valves makes use of the higher pressure of the accumulator compared to port B to cause the control valve to open rapidly. The port cuts off about 95% of travel to cushion valve travel. The remaining oil goes through small port or a groove. This action all occurs with the operator's forward acceleration signal.

3. Vehicle coasting, some accumulator charge, operator signals regenerative braking.

There is no pilot valve signal, but instead pressure build-up at port B forces valve open, oil over valve will port to A via passage 1. Once open, the valve remains open unless signaled to close, and only if the pressure at A and B is at least 600 psi above R, the reservoir.

4. System discharged.

The spring will open the valve, readying the vehicle for pedaling via A or regenerative braking via B.

Figure 12 illustrates the operation of valve 23 of Figure 2. Figure 13 is a schematic diagram of the valve 24 of Figure 2. Referring to Figure 2, it will be noted that the wheel motors 30 are shown with ports X and Y which are connected to X and Y of valve 23. As illustrated the valve also includes ports 1, 2, 3 and 4 and a connection leading to the reservoir 21. The valve is shown in the rest or motor position. When the valve moves from the rest position to the non-rest position it redirects the motor port X from port 2 to port 1 and blocks off the Y motor port from port 3. The Y motor port can still exit to port 4 through a check valve. Similarly with valve 24 of Figure 13, the valve is shown in the open position, not the rest position and moving the spool to the rest position blocks off the pedal pump P port from the accumulator circuit.

Figure 14 is a schematic diagram of the construction of the hydraulic variable pressure ratio transducer 25 of Figure 2. As illustrated the transducer consists of an ellipsoid cam ring 90, a freely rotating armature 91, a porting ring 92 and a control vane 93. The end plates of the transducer are connected as shown in Figure 21 so that an external port 94 is connected to the wheel motors. Ports 95 and 96 are connected to the accumulator and the central opening in control vane 93, port 97 connects to the return to the reservoir.

As detailed in Figures 19 and 20, the armature 91 which rotates freely about the porting ring 92 is provided with a plurality of radial vanes and radial holes positioned between the vanes for communication with the interior of the porting ring 92.

Figure 15 is an axial view of the porting ring 92 showing ports 95 and 96 for connection to the accumulator and illustrating openings 99 and 100.

Figure 16 is a view of the porting ring 92 at right angles to Figure 15 showing the port 99 in phantom, the port 100 and the connections from the ports 95 and 96 to the opening 99 and 100 in phantom.

Figure 17 illustrates the control vane 93 with control vane oil entrance and exit 97 to the reservoir. Additionally there are shown oil ports 101 and 102 which are positioned in openings 99 and 100 of the porting ring 92 and receive oil from the radial holes in the armature 91 as detailed in Figures 19 and 20.

Figure 18 illustrates a view at right angles to Figure 17 of the control valve 93 showing the control shaft 103 which is connected by means not shown to a control mechanism not illustrated for varying the position of the vane and thereby varying the pressure ratio of the hydraulic variable ratio transducer 25.

Figure 19 is an axial view of the armature 91 which includes a plurality of slots 110 carrying dual pressurized vanes 111. In accordance with standard high pressure vane technology, the vanes 111 can be pressurized from the bulbous portion at the inner end of each slot 110 to cause the vanes to extend to the cam ring 90 as shown in Figure 14. The armature includes radially drilled holes 112 between adjacent pairs of vanes.

It would be appreciated by those skilled in the art that the control vane is mounted inside the porting ring which in

turn is mounted inside the freely rotating armature, and in turn the armature being mounted inside the cam ring is all positioned between end plates, one of which permits the shaft 103 to extend external to the transducer and the other of which is illustrated in Figure 21. The porting ring 92 is fixed immobile between the end plates, with ports 95 and 96 aligned with end plate connections 105 and 106. Port 94 is of course aligned with opening 104 in end plate 115, and opening 107 is in line with opening 97 of control vane 93. Port 104A is also provided in end plate 115 and is connected to port 104 and the transducer is therefore a two chamber mechanism. The characteristics of the transducer are that it is a variable flow constant pressure ratio device, the ratio being set by the position of the control vane 93 under the control of the operator.

This is in contrast to a flow control valve which is a constant volume device that throttles the flow, all of the throttled energy being wasted. The transducer is relatively simple having one rotor, one control valve shaft, two chambers, three ports and twelve vanes in the embodiment illustrated in Figure 19. Since there is no throttling of the flow the energy efficiency of the device is extremely high, there being virtually no loss in the transducer.

Figure 22 illustrates a four chamber version of the transducer, with the armature removed for clarity. As before, the armature consists of slotted vanes with radial holes drilled between each pair of vanes similar to the armature of Figures 19 and 20.

Figure 23 illustrates the center control porting ring and the rotary control section to control pressure area of accumulator section.

The transducer as illustrated in Figures 14 through 21 inclusive is a hydraulic variable transformer that performs this function as a free turning armature with no drive shaft.

The armature is a standard vane type as seen in vane pumps and motors commonly available, except that a hole has been drilled radially in between each pair of vanes to the center.

The transducer is ported conventionally in zone A illustrated in Figure 14, which is connected only to the wheel motor and pedal pump circuit and allows oil in and out of zone A. The armature rides on the porting ring which is stationary at all times and has an oil entrance-exit at its stationary vanes. The control vane can rotate approximately 90° and its position determines the pressure ratio between zone A and zone B. The control vane position is controlled through the shaft 103 as illustrated in Figure 18.

The armature 91 will rotate as fast as necessary to cause the pressure balance to be re-established and can reverse direction. Direction reversal is invisible to the operator. As the armature rotates, oil in zone A cannot exit via radial holes since they are blocked by parts of the stationary porting ring. Once in zone B or zone C, oil can only exit or enter via radially drilled armature holes because the end porting is not present in these areas. There is a reaction torque on the control vane which can be used for control feedback to the operator.

A four chamber variation of the transducer is illustrated in Figures 22 and 23 and accomplishes the same task as the two chamber version of Figures 14 through 21. The rotary control section E of Figure 23 ports oil through radial armature holes to effect a pressure distribution in the cam ring. Oil which flows out of zone A via radial holes would return via radial holes from area F to zone B when the armature is turning counterclockwise. If the armature is turning clockwise, zone A oil enters zone X and then exits via end ports C in the end plate. Stationary segment D of the porting ring prevents oil from returning via radial holes in zones X and Y.

If rotary segment E is rotated counterclockwise from the position shown in drawing, then zone A would also move counterclockwise in the cam ring. Zone B being counterclockwise from zone A would shrink while a new zone B clockwise from zone A would develop. This has the effect of reducing displacement of zone A. When zone A is centered around line P of Figure 22, it has zero displacement. Continuing to move zone A counterclockwise will again increase displacement but in the opposite direction. Eventually zone A and zone B will have exchanged positions. If the transducer is in a position as shown, accumulator oil could exit into zone A causing armature to rotate clockwise and pressurize zone X, delivering power through port C. Motor return oil would enter in port C in zone Y and exits zone B through radial holes to area F (Figure 23). Oil could also be delivered to zone X by a pump causing counterclockwise armature rotation delivering oil to accumulator via zone A. Oil would be entering zone B from the reservoir and returning via zone Y back to the reservoir along with any oil exiting the motor if present. The transducer would change rotational direction as needed, and this would be invisible to the operator.

Rotating control section E counterclockwise to the area shown on zone B would allow zone Y oil to cause a clockwise rotation charging the accumulator.

As with the mechanism of Figures 14 to 21, there is a finite number of vanes and some cogging action may result in certain conditions, particularly at low delivery. A small fast rotating design would, of course, reduce this effect.

The hydraulic circuit of a vehicle in which this transducer of Figures 22 and 23 would be used is required to be modified for this transducer. The main difference is that the rotary control section reverses pressure direction so that a motor reversing valve would not be required in a typical hydraulically actuated vehicle.

Since rotary control section E is in hydraulic balance, both radially and circumferentially, there are therefore no control forces. Operator feedback is therefore not available but could be supplied by a spring or other similar device.

Area F in Figure 23 is connected to the reservoir and changes in size as needed. There would be four area F's if the rotary control section is not at an end travel point. Area F always has a corresponding zone B in the cam ring. With more high and low pressure interfaces than the device of Figures 14 to 21, there would be more hydraulic leakage losses when compared to that device in the mechanism of Figures 14 to 21.

As seen from the explanations of the two embodiments, the transducers are two basic approaches to the design, and it is important to realize that these two designs can each have other numbers of chambers. The difference in approach is really of a three port device and four port device.